Georgia Institute of Technology
Atlanta, Georgia 30332

ACTIVELY CONTROLLED SHAFT SEALS FOR AEROSPACE APPLICATIONS

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Principal Investigator Richard F. Salant
College of Engineering
School of Mechanical Engineering

NASA Technical Officer: M.P. Proctor
NASA Lewis Research Center, MS SPT D-2
Space Vehicle Propulsion Branch

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1. OBJECTIVE AND SUMMARY

The objective of years 4 and 5 of this project (1992 and 1993) is to determine experimentally the behavior and operating characteristics of a controllable mechanical seal, and to identify potential problem areas. A controllable mechanical seal is one in which the thickness of the lubricating film separating the sealing surfaces is adjustable, and can be controlled by an electronic control system, based on information supplied by sensors that monitor the condition of the film.

This work builds upon work done during years 1 -3, in which a controllable mechanical seal was designed, analyzed and fabricated. At the beginning of year 4, the mechanical seal and test rig had been assembled, and preliminary testing had begun.

2. RESULTS OF GRANT TO DATE

The five major tasks of years 4 and 5 encompass instrumentation, configuration changes of the mechanical seal to optimize its performance, systematic steady state tests, systematic transient tests, and a final report. During this reporting period, significant progress has been made on instrumenting the test rig and modifying the design to optimize the seal's performance. Initial steady state tests have also been performed.

Task 1. Instrumentation

A. Face Temperature

The conditions of the lubricating film are monitored in terms of the face temperature. The latter is measured by thermocouples, which are located in the carbon faces on each side of the seal. The thermocouples have been installed and initial steady state tests indicate that they are working properly (see Steady State Tests).

B. Leakage Rate

The helium inlet to the seal has been instrumented with a rotameter. The rotameter was field calibrated by collecting the helium in an inverted beaker for specified amounts of time. The rotameter is capable of measuring leakage rates up to 3500 ml/min. It may be necessary to install a flowmeter capable of measuring a larger range of leakage rates (see Steady State Tests).

C. Pressure

The pressure of the helium entering the seal is measured with a pressure gage in line with the helium flowline. The pressure gage has been installed and is working properly.

D. Voltage

The voltage applied to the piezoelectric element is currently measured with a volt meter located on the front panel of the power supply. The volt meter on the power supply was found to be in good agreement with a hand held voltmeter. Two additional power supplies are currently being purchased, which can be interfaced with a data acquisition card to give a remote readout of voltage.

E. Rotational Speed

The rotational speed is currently being measured with a small (1.5-3.0 volt) D.C. motor, which is attached to the shaft of the test rig. The D.C. motor acts as a generator and puts out a voltage proportional to the rotational speed of the test rig. The motor was calibrated with a handheld optical tachometer.

F. Data Acquisition

A PC-based data acquisition card has been acquired. This will be used to continuously monitor the face temperature, voltage applied to the piezoelectric element and rotational speed.

Task 2. Configuration Changes

A. Electrical Insulation

At the completion of year 3, static pressure tests were performed with the test rig, and it was found that the seal leaked through the o-rings located between the deformable face assembly (the carbon face and piezoelectric element) and holder (see Figure 1). This was caused by the large tolerances produced by the epoxy coating on the outside radius of the piezoelectric element. The epoxy coating was necessary to prevent electrical breakdown between the piezoelectric element and the stainless steel holder.

The holders were remade from a nonconductive plastic, acetyl copolymer, which is commonly referred to by its tradename delrin. This material was not intended for use in the final design, but rather was an economical design change to demonstrate proof of principle, that the leakage in the seal can be controlled by adjusting the voltage applied to the deformable face assembly. This eliminated the need for the epoxy coating on the deformable face assembly and provided a smooth sealing surface on the outside radius of the piezoelectric element. Plastic bushings were also machined and placed on the deformable face assembly to prevent electrical breakdown between the inside radius of the piezoelectric element and the steel shaft (see Figure 1).

When the new holders were in place the seal was again pressure tested and was found not to leak. In addition the deformable face assemblies sustained voltage levels up to 3000 volts with no electrical breakdown. Steady state tests were performed (see Steady State Tests), and it was found that the leakage in the seal could be controlled by adjusting the voltage applied to the deformable face assembly. However, a major disadvantage of the plastic holders was that they have very low thermal conductivity, which causes the seal faces to retain heat.

To remedy this, the holders are being remade from combat boron nitride, which has the properties of high dielectric strength and high thermal conductivity. This will allow the deformable face assembly to sustain high voltages while providing better heat dissipation. The new holders are currently being fabricacated.

B. Balance Ratio and Spring Force

The balance ratio for the initial design was 0.65, which was somewhat low. With this balance ratio a relatively large spring force (provided with wave spring washers) was required to produce the necessary closing force. A large spring force is not desirable because of the high manufacturing tolerances for springs. These tolerances could lead to large changes in the closing force, which would adversely affect the performance to the seal.

To address this, the balance ratio has been changed to 0.75 and the wave springs have been replaced by six coil springs for each side of the seal. With the larger balance ratio, the spring force is significantly reduced, which will accordingly reduce the variation in the closing force produced by the manufacturing tolerances in the springs.

C. Piezoelectric Geometry and Deformation Mode

Bench tests were performed with the deformable face assembly to determine the amount of coning produced by the voltage applied to the piezoelectric element. Figure 2 presents results from one of these tests. The range of coning produced was from -2.5 to +3 microns for a voltage range of -2500 to +2000 volts. This range of coning should be sufficient to provide the required range of film thickness for the controllable seal.

Task 3. Systematic Steady State Tests

Systematic steady state tests were initiated to determine the leakage in the seal as a function of the voltage applied to the deformable face assembly. Static tests were performed while the seal was not rotating and dynamic tests were performed while it was rotating at approximately 20,000 rpm.

Figure 3 presents leakage versus voltage for the static test. This figure indicates that the leakage can be controlled by adjusting the voltage applied to the deformable face assembly. The range of leakage obtained in this test was from 0 ml/min to leakages greater than 3500 ml/min. The maximum leakage that can be measured by the rotameter is 3500 ml/min.

Figure 4 presents film thickness versus voltage in which the film thicknesses correspond to the leakage rates in Figure 3. To compute the film thickness it was assumed that each side of the seal had the same film thickness and that the nondimensional coning was 0.25. This figure indicates that film thicknesses can be obtained from 0 um to greater than 3.0 um.

Figure 5 presents leakage versus voltage for the dynamic test. The rotational speed was approximately 20,000 rpm. This figure indicates that the leakage in the seal can be controlled by adjusting the voltage applied to the deformable face assembly while the seal is rotating. Figure 6 presents film thickness versus voltage for the rotating test. This figure shows that the lubricating film can be adjusted to values larger than 3 um (however the rotameter goes off scale for film thicknesses greater than 3 um). The lower limit for the film thickness was approximately 2.2 um. The seal was not adjusted to run at lower film thicknesses because of the high temperatures, which were produced due to the low thermal conductivity of the plastic holder.

Figure 7 presents the temperature of the seal versus time. Also presented on the right-hand axis is the film thickness. The face temperature continually increased during the test with no major reduction in temperature as the film thickness was increased. This is presumed to be due to the poor thermal conductivity of the plastic holder. To address this the holders are being remade from a thermally conductive ceramic (see Configuration Changes).

3. PRESENTATIONS/PUBLICATIONS

During this reporting period the following presentations/publications were generated:

Wolf,P. and Salant, R.F., "Design and Analysis of a Controllable Mechanical Seal for Aerospace Applications," Fourth International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Honolulu, Hawaii, 1992.

Salant, R.F., "Maintaining Full Film Lubrication in Mechanical Seals by Means of Electronic Control," Tribologia, Finnish Journal of Tribology, vol. 11, no. 2, pp. 122-129, 1992, and Fifth Nordic Symposium on Tribology, Helsinki, Finland, 1992.

Copies of these papers are attached to this report.

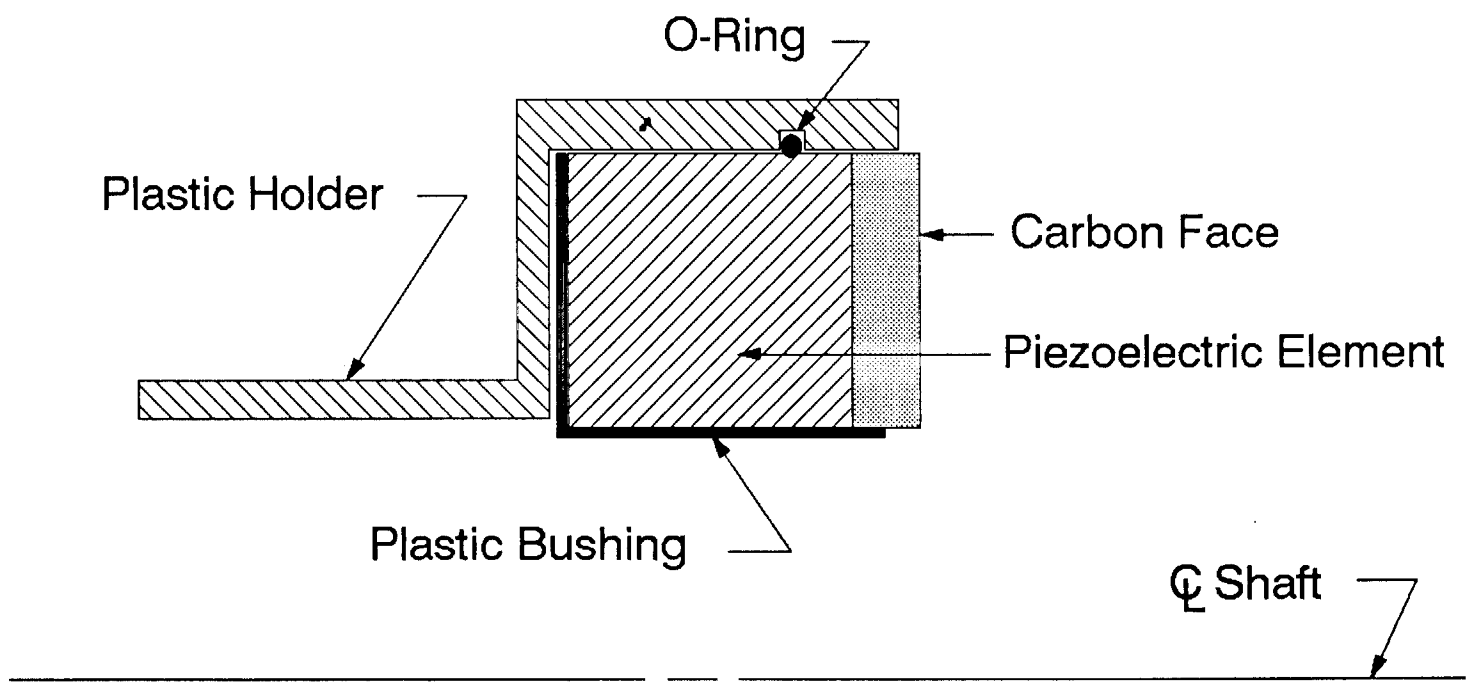

Figure 1. Holder and Deformable Face Assembly

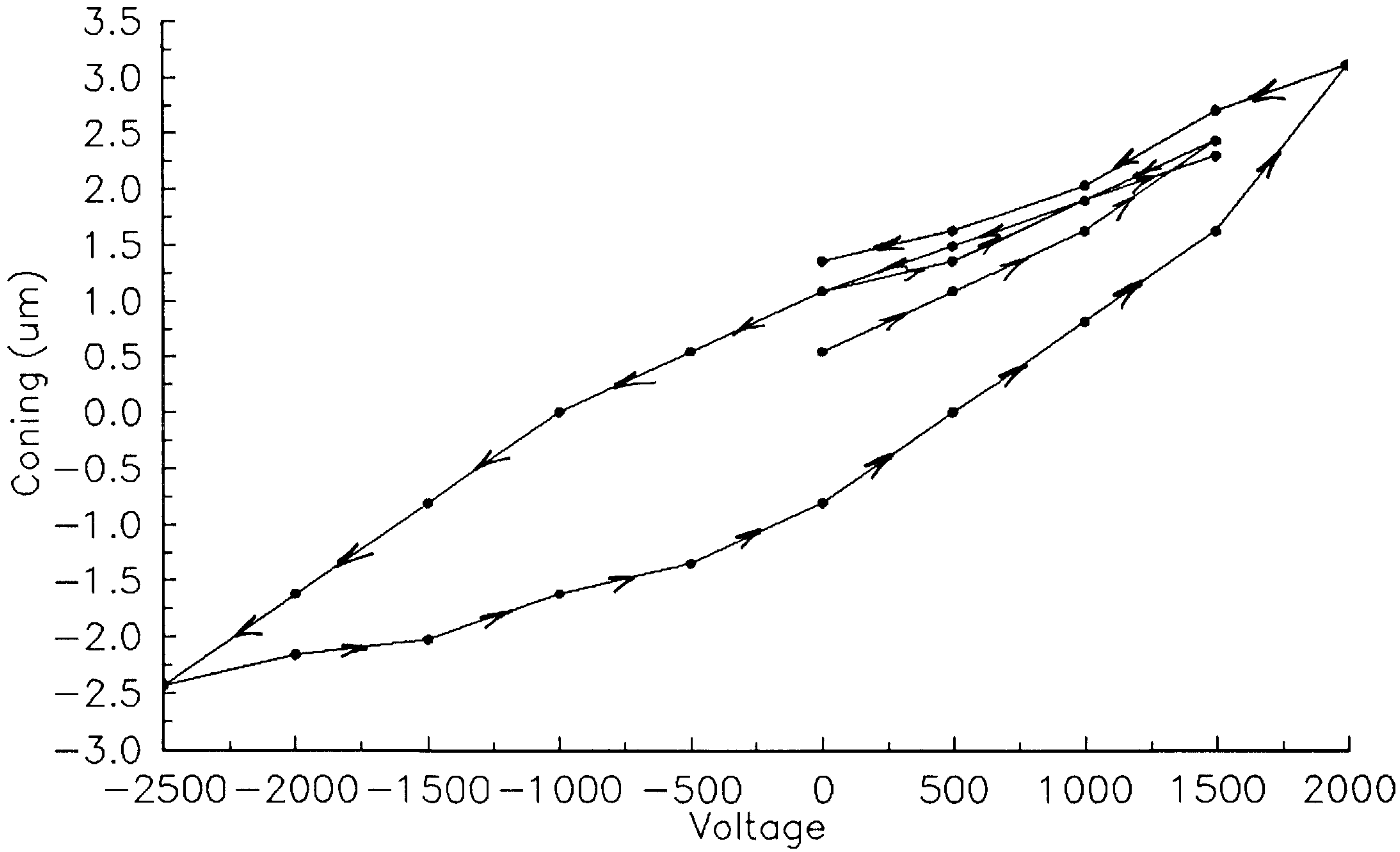

Figure 2. Coning versus Voltage

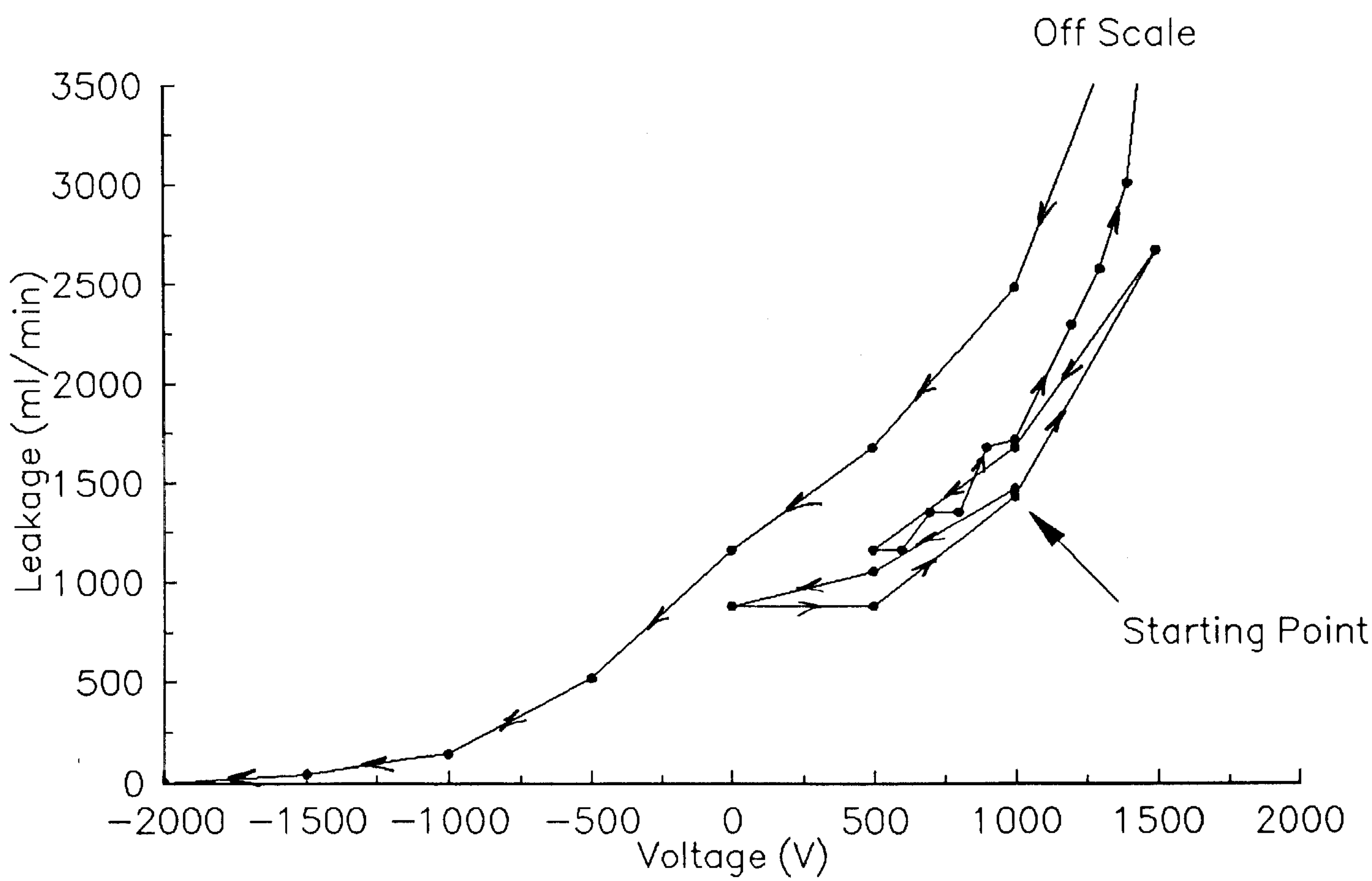

Figure 3. Leakage versus Voltage — Static Test

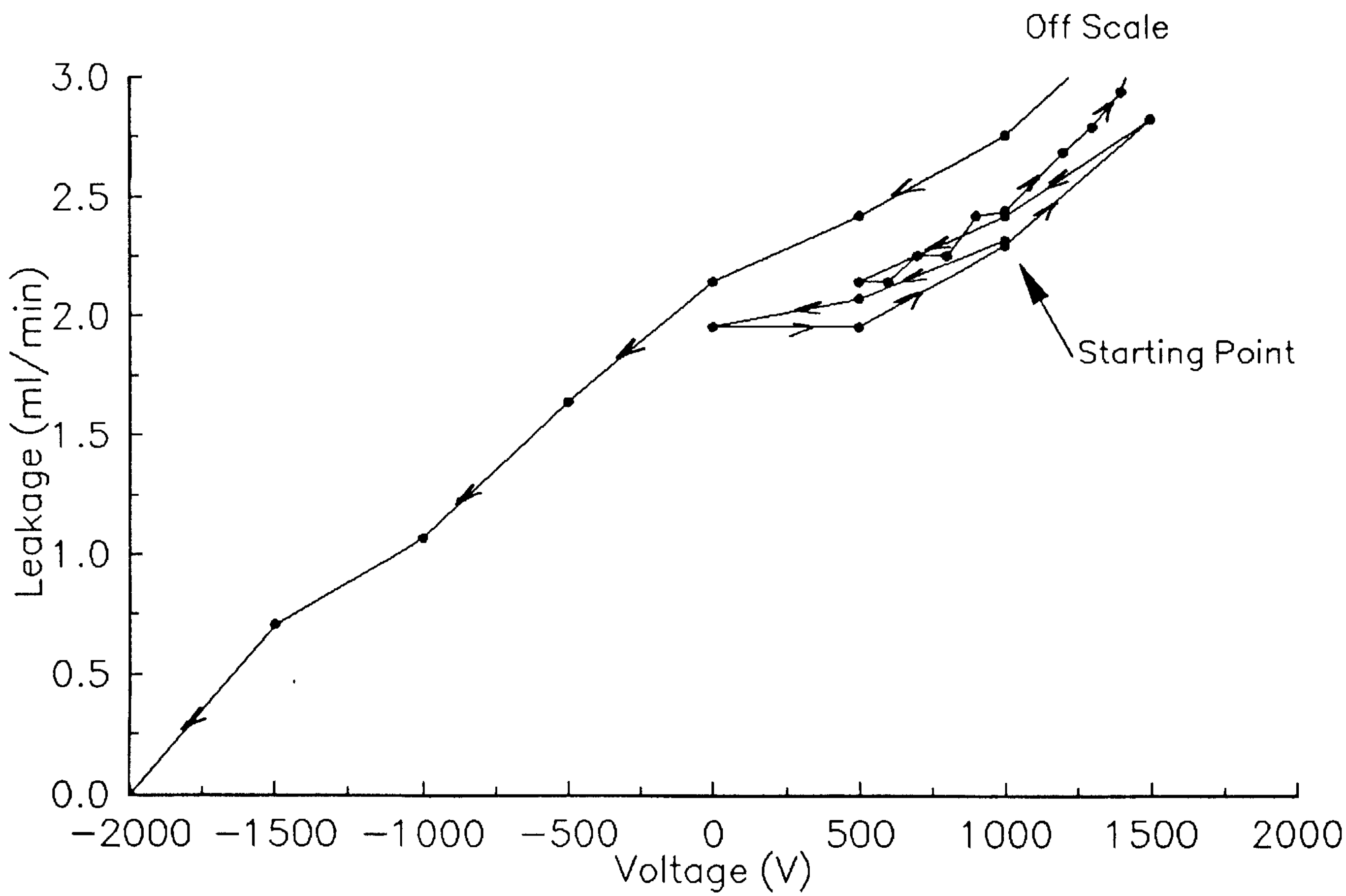

Figure 4. Film Thickness versus Voltage – Static Test

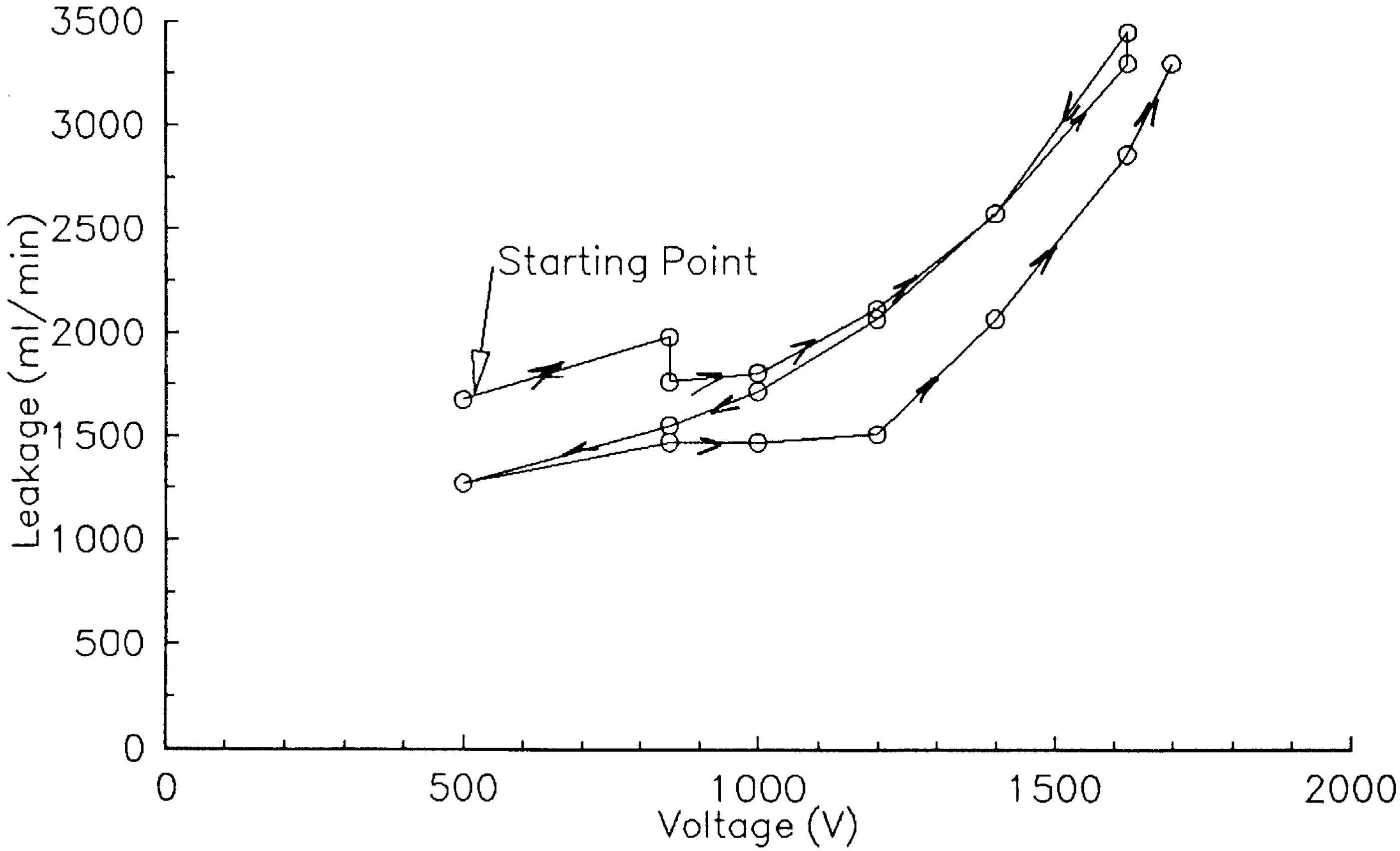

Figure 5. Leakage versus Voltage — Rotating Test

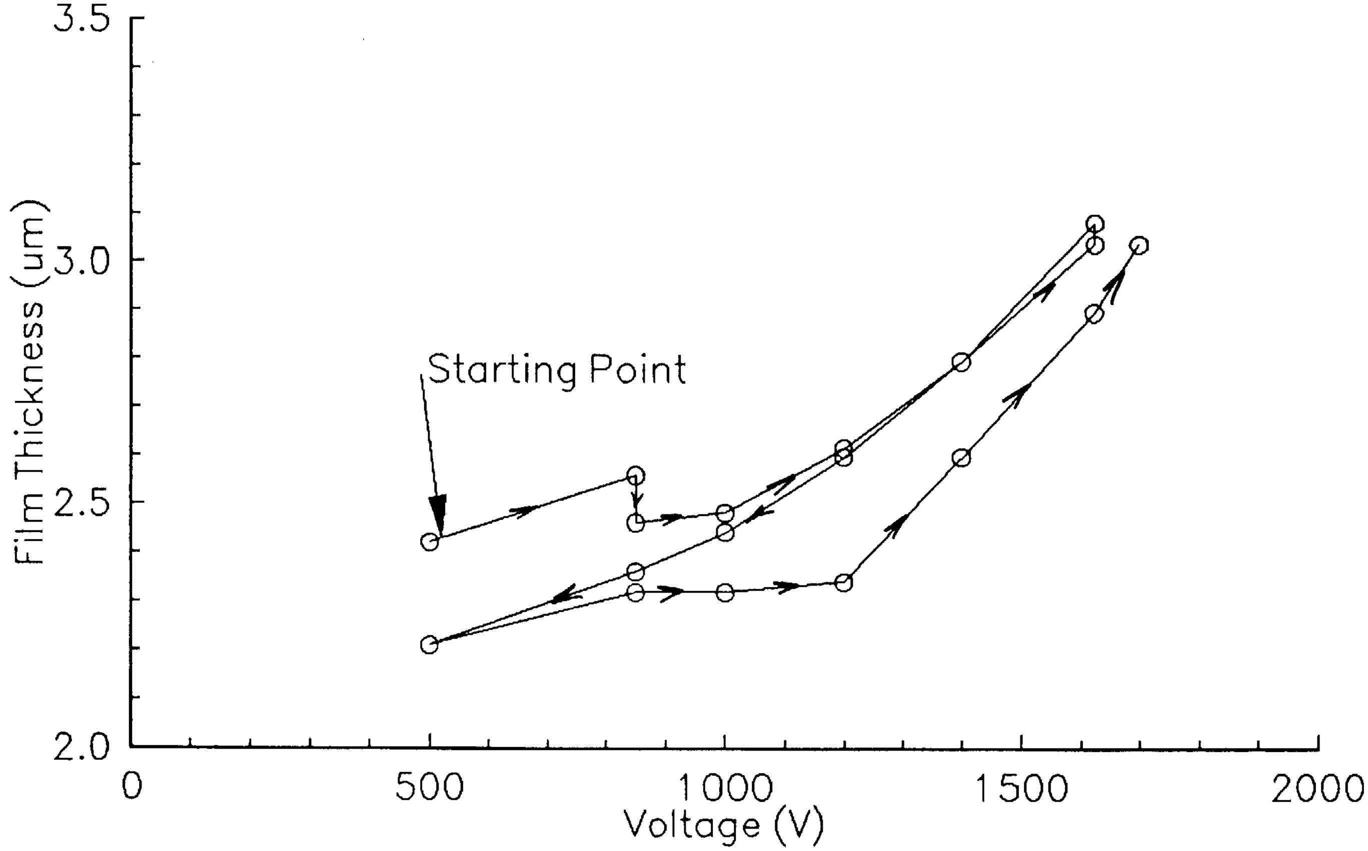

Figure 6. Film Thickness versus Voltage – Rotating Test

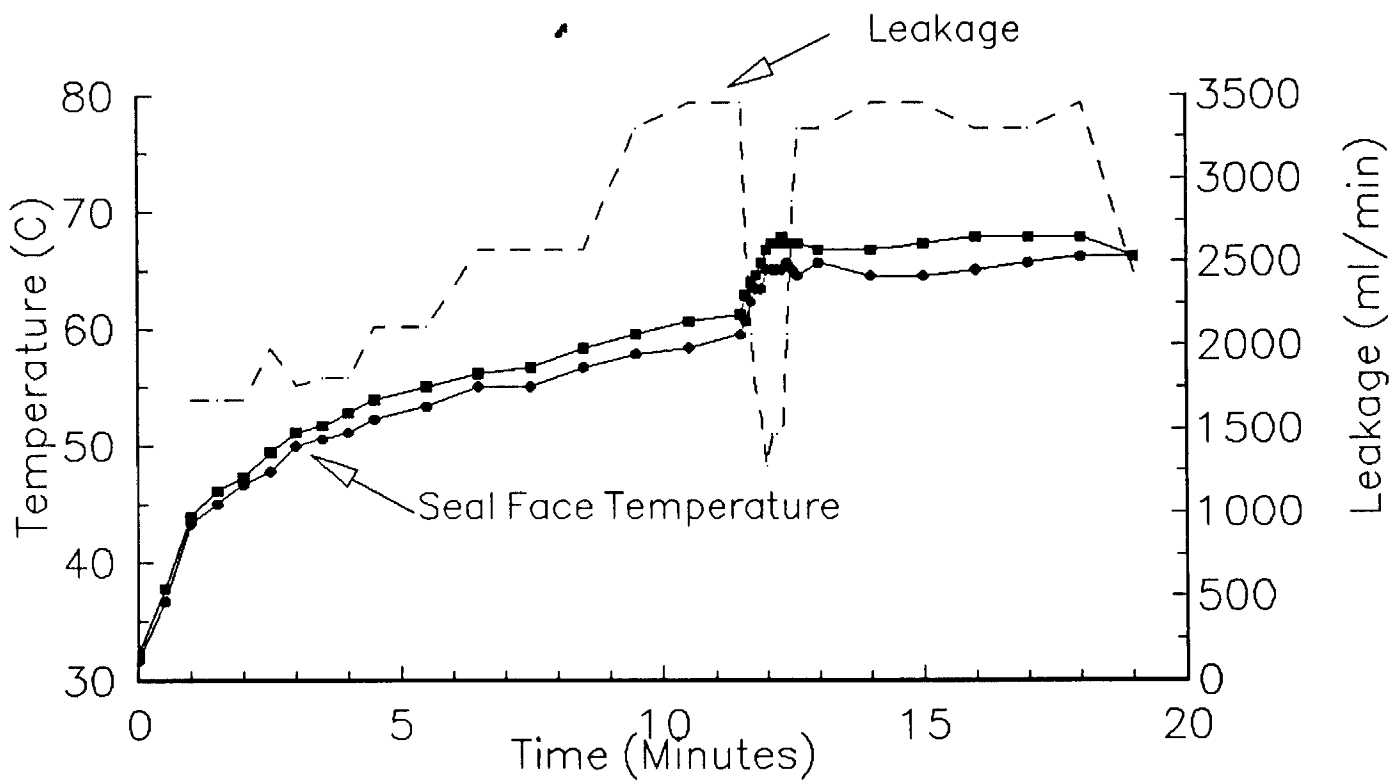

Figure 7. Seal Face Temperature versus Time

DESIGN AND ANALYSIS OF A CONTROLLABLE MECHANICAL SEAL FOR AEROSPACE APPLICATIONS

Paul Wolff and Richard F. Salant
School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, Ga. 30332

ABSTRACT

A controllable mechanical seal for aerospace applications has been designed and analyzed. Active control of the film thickness is achieved by controlling the seal face coning with a piezoelectric actuator. An analytical model has been used to evaluate the effects of the closing force, thermal boundary conditions, pressure variations, piezoelectric material properties, and premachined coning, on the performance of the seal. The results generated by this model indicate that an actively controlled mechanical seal for aerospace applications is feasible.

INTRODUCTION

Although mechanical seals have very low leakage rates, their use in the aerospace industry is limited, due to their reduced reliability as compared to fixed clearance seals. The reduced reliability is caused by repeated incidents of face contact, resulting from a breakdown of the lubricating film that separates the seal faces.

To increase reliability, the actively controlled mechanical seal has been developed (Salant, et. al., 1989). Control of the film thickness is achieved by incorporating a piezoelectric actuator in one of the seal components. The actuator alters the geometry of the seal interface (coning), which regulates the pressure distribution within the lubricating film. The pressure distribution, in turn, determines the opening force acting on the floating face, which controls the film thickness; increasing the opening force increases the film thickness. The film thickness is monitored by a thermocouple that is embedded in one of the seal faces. An optimal film thickness is continually maintained with such a controllable mechanical seal, through suitable adjustment of the voltage applied to the actuator.

Such actively controlled seals have been previously developed for industrial use. However, extending this design to aerospace applications presents a challenge due to the stringent aerospace design constraints. In particular, an aerospace seal must be much smaller than an

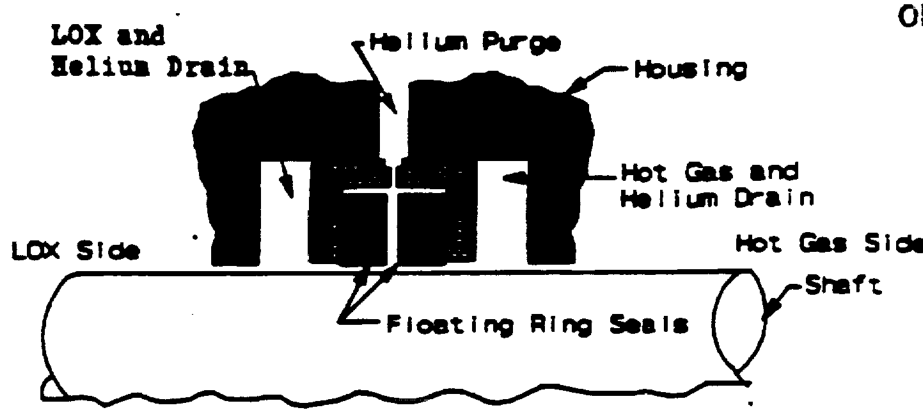

Figure 1. Helium Purge Assembly

maintaining a minimal film thickness. If, at any time, the onset of face contact is detected, the voltage is immediately increased to prevent such contact. At a later time, the voltage is again decreased to reduce leakage.

DESIGN CONSTRAINTS

In a LOX turbopump, contact between the LOX and hot turbine gases is prevented with a series of seals which includes the helium purge assembly, Figure 1. Pressurized helium is introduced near the midpoint of the turbopump. The high pressure helium prevents contact between the lower pressure LOX and hot gases. Currently, floating ring seals, having relatively high leakage rates, are utilized in the helium purge assembly, as shown in Figure 1. The object of this project is to replace those seals with controllable mechanical seals to reduce helium leakage and maintain high reliability.

The size and operating conditions of a LOX turbopump impose severe constraints on the design of a mechanical seal. The dimensions of the seal envelope are 3.81 cm in the axial direction and 1.27 cm from the outside radius of the shaft to the outside radius of the envelope. These dimensions are considerably smaller than those of previous actively controlled seals. The high and low pressure sides of the seal are at 1.38×10^6 Pa and 1.10×10^5 Pa, respectively. Temperatures that affect the seal design include the temperature of the hot gas and helium drain (469 K), the temperature of the LOX and helium drain (123 K) and the temperature of the helium on the high pressure side (294 K). The thermal deformations produced by these temperature gradients have a significant effect on the coning deformations of the seal faces and influence the performance of the seal. The shaft of the turbopump rotates at a speed of 7,330 rad/sec.

SEAL CONFIGURATION

A double mechanical seal configuration has been chosen to replace the double floating ring seals, as shown in Figure 2. It is comprised of one rotor and two nonrotating floating components. Each floating component consists of a deformable face assembly, a holder, a wave spring washer and two o-rings. The final design has been developed through an iterative process of design, analysis (see next section), and redesign.

MATHEMATICAL MODEL

The mathematical model used to evaluate various seal designs is based on an approach utilized by previous researchers (Salant and Key, 1984) and consists of four major elements: a force balance, fluid mechanics model, heat transfer and structural model, and an overall computational procedure. The primary outputs of concern are the film thickness and leakage rate for the range of voltage applied to the piezoelectric element. This model assumes steady state operation of the seal.

The first step in computing the film thickness is to perform a force balance on the floating component ($F_{open} = F_{close}$). Computation of the closing force is straightforward (see Equation 1). However, to determine the opening force, the pressure distribution within the fluid film must be computed from the governing fluid mechanics equations.

The pressure distribution is governed by the Navier-Stokes Equations. An analytical expression can be obtained from these equations when the following simplifying assumptions are made (Hughes et. al., 1989; Wolff, 1991): steady state, axisymmetric fluid film, isothermal fluid film, ideal gas behavior, laminar flow, no centrifugal effects, no velocity gradients in the radial direction, and no squeeze film effects. Using these assumptions, a nondimensional equation for the pressure distribution is obtained,

$$
\begin{aligned}
P^* = [\ln r^* - \ln(1+\delta^*\alpha^*(r^*-1)) + (1-\delta^*\alpha^*)\left(\frac{1}{1+\delta^*\alpha^*(r^*-1)}-1\right) \\
+ \frac{(1-\delta^*\alpha^*)^2}{2}\left(\frac{1}{[1+\delta^*\alpha^*(r^*-1)]^2}-1\right)] / [\ln r_o^* - \ln(1+\delta^*\alpha^*(r_o^*-1)) \\
+ (1-\delta^*\alpha^*)\left(\frac{1}{1+\delta^*\alpha^*(r_o^*-1)}-1\right) + \frac{(1-\delta^*\alpha^*)^2}{2}\left(\frac{1}{[1+\delta^*\alpha^*(r_o^*-1)]^2}-1\right)]
\end{aligned}
\tag{3}
$$

Once the geometry (α^*, r_o^*) of the seal is specified, the pressure distribution depends only on δ^*, the nondimensional coning. As δ^* increases, the pressure distribution becomes more convex, which results in a larger opening force. For a given closing force, there is a unique value of δ^* (δ/h_i) that produces an equal opening force. Once this value is found, it is necessary to compute the dimensional coning δ, to find the steady state film thickness h_i.

To compute δ, one must determine the deformations of the seal components due to pressure loading, thermal stresses, and voltage applied to the piezoelectric actuator. This is done using the finite element code ANSYS. As input to the finite element program, the viscous heat generation rate in the fluid film is expressed as,

$$
\dot{Q} = \int \frac{\mu \omega^2 r^2}{h} dV \tag{4}
$$

where a linear radial profile is assumed for h. Once the viscous heat generation rate is computed, it is apportioned between the seal faces based on their relative thermal conductivity. In addition to the viscous heat generation rate, the thermal boundary conditions also must be supplied to the finite element model, to compute the temperature distributions of the seal components. Each floating component experiences different thermal boundary conditions of film thickness and seal component temperatures are compared to the old values. If the deviations are within specified tolerances, the solution has converged; if not, the procedure is repeated until convergence is achieved. Five voltage loads are supplied for each computational run: 6000, 3000, 1000, 500, and 100 volts. The results obtained from each run include the film thickness, leakage, coning, temperature distributions of the seal components, and pressure profile within the fluid film.

Once the film thickness is known, the leakage rate may be obtained by integrating the pressure gradient at the seal ID, across the thickness of the film. Assuming a linear radial profile for h, an analytical expression for the leakage rate is obtained,

$$\dot{m}=[-\pi(P_o^2-P_i^2)(h_o-\beta r_o)^3]/12\mu RT[\ln(\frac{r_o}{r_i})-\ln(\frac{h_o}{h_i}) + (h_o-\beta r_o)(\frac{1}{h_o}-\frac{1}{h_i})+\frac{(h_o-\beta r_o)^2}{2}(\frac{1}{h_o^2}-\frac{1}{h_i^2})] \tag{5}$$

Two additional parameters of concern are the stiffness and controllability of the seal. The stiffness is important to the stability of the seal, and is defined by,

$$k=-\frac{dF_{open}}{dh_i}=\frac{dF_{open}}{d\delta^*}\frac{\delta^{*2}}{\delta} \tag{6}$$

The stiffness must be positive for stable seal operation, and also must be large enough to prevent large fluctuations in the film thickness due to small changes in the closing force. For fixed values of $dF_{open}/d\delta^*$ and δ, the seal stiffness increases as δ^* increases.

The controllability of the seal is a measure of the sensitivity of the film thickness to changes in voltage, and is defined as,

$$\frac{dh_i}{dv}=\frac{dh_i}{d\delta}\frac{d\delta}{dv}=\frac{1}{\delta^*}\frac{d\delta}{dv} \tag{7}$$

A large controllability is desirable, to minimize required voltage levels. Equation (7) indicates that the controllability of the seal varies directly with $d\delta/dv$, and inversely with δ^*.

As described above, δ^* strongly affects both the stiffness and controllability of the seal. However, δ^* is determined by the closing force. Therefore, the latter plays a major role in determining the stability and controllability of the seal.

RESULTS

The stability and controllability of the seal are dependent on δ^*, which is uniquely determined by the closing force. Therefore, it is essential to determine the optimum value of the closing force.

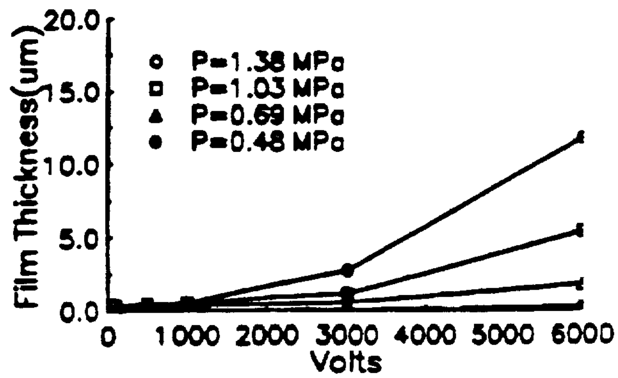

Figure 6. Seal Performance - Variable Pressure Operation

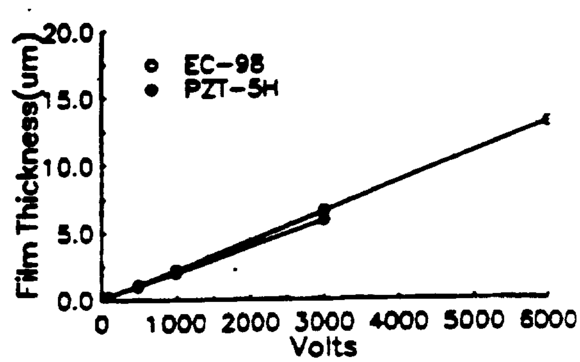

Figure 7. Seal Performance - Comparison of Piezoelectric Materials

CONCLUSIONS

The results presented in this study indicate that an actively controlled seal for the helium purge assembly of a LOX turbopump is feasible. Reasonable film thicknesses, on the order of a few microns, can be obtained with the proposed design. The range over which the film thickness can be varied, by applying voltage to the piezoelectric actuator, is as large as twelve microns.

The next phase of this study, involving fabrication and laboratory tests, is now underway.

ACKNOWLEDGMENT

This work has been supported by the National Aeronautics and Space Administration, Lewis Research Center, under Grant NAG-3-974. The NASA Technical Officer has been M.P. Proctor.

NOMENCLATURE

A_{face}	area of seal face
d_i	axial deformation of seal faces at inside radius
d_o	axial deformation of seal faces at outside radius
F_{close}	closing force
F_{open}	opening force
F_{spring}	spring force
h_i	film thickness at inside radius
h_o	film thickness at outside radius
k	seal stiffness
m	mass flow rate
N_b	balance ratio
P	pressure
P_i	pressure at inside radius of seal
P_o	pressure at outside radius of seal
P^*	nondimensional pressure $(P^2-P_i^2)/(P_o^2-P_i^2)$
r	radial coordinate
r_i	inside radius

FOURTH INTERNATIONAL SYMPOSIUM ON TRANSPORT PHENOMENA AND DYNAMICS OF ROTATING MACHINERY (ISROMAC-4)

VOLUME A

APRIL 5-8, 1992

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PREPRINTS

RICHARD F. SALANT, PAUL WOLFF & SAMUEL NAVON
Georgia Institute of Technology
School of Mechanical Engineering
Atlanta, Georgia 30332
U.S.A.
Tel. (404) 894-3176
Fax (404) 894-8336

MAINTAINING FULL FILM LUBRICATION IN MECHANICAL SEALS BY MEANS OF ELECTRONIC CONTROL

INTRODUCTION

Most modern mechanical seals are designed to operate with full film lubrication between the mating faces, under steady state conditions at the design point. The presence of a thin lubricating film, with a thickness on the order of a few microns, minimizes wear and the occurrence of mechanical and thermal damage to the faces. However, under transient and off-design conditions the lubricating film frequently breaks down, resulting in mechanical contact between the faces. To prevent such face contact, a new class of "controllable" mechanical seals is under study and development by several researchers [1-4].

PRINCIPAL OF OPERATION

A typical conventional mechanical seal is shown schematically in figure 1. The rotating face is floating, and is free to move in the axial direction (within limits), while the nonrotating face is fixed axially. The thickness of the lubricating film that separates the faces is determined by the axial location of the floating face, which is governed by the film thickness, which can be adjusted explicitly. If conditions are such that the film is too thick or too thin, it is readjusted to its optimum thickness while the seal is in service. The film thickness could be controlled by controlling either the closing force or the opening force. While some researchers control the former [2,4], the present authors have found it is much more effective to control the latter. Since the opening force and the film thickness are strongly dependent on coning, control is achieved by controlling the coning. This is accomplished by incorporating within the seal a piezoelectric actuator to deform mechanically one of the faces and generate coning.

INDUSTRIAL AND AEROSPACE SEALS

Based on the concept described above, several controllable seals for industrial applications (e.g., feedwater pumps) have been built and successfully tested. One such seal is schematically illustrated in figure 2, which shows a nominal 50 mm water seal with faces constructed of stellite (rotating face) and carbon graphite (nonrotating face). The configuration is similar to that of a conventional seal, except the nonrotating face holder has been modified to allow a piezoelectric actuator to be placed behind the backside of the face. When a positive voltage is applied to the actuator, the piezoelectric elements within the actuator expand, and exert a force on the backside of the face. Since the outer diameter of the face is constrained from axial movement by the holder, this force produces face deformation contributing to positive coning. The larger the applied voltage, the larger the coning and the thicker the fluid film. Thus, by adjusting the voltage, one can adjust the film thickness.

To determine the voltage to be applied to the actuator for optimum operation at any given instant of time, information on the conditions in the fluid film and the sealed cavity is required. These conditions are monitored by a thermocouple imbedded in the nonrotating seal face, and a second thermocouple in the sealed cavity. The outputs of these thermocouples are transmitted to an electronic adaptive control system, where they are processed and used to generate a commanded actuator voltage, as specified by an adaptive control strategy. Such a strategy searches for and maintains an optimum

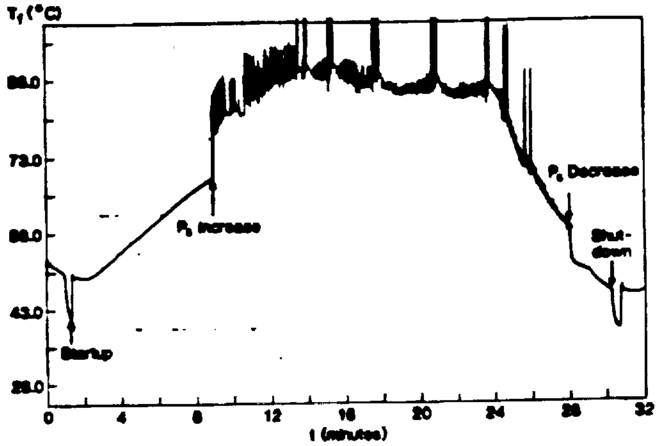

FIGURE 3. Transient performance, conventional seal [1].

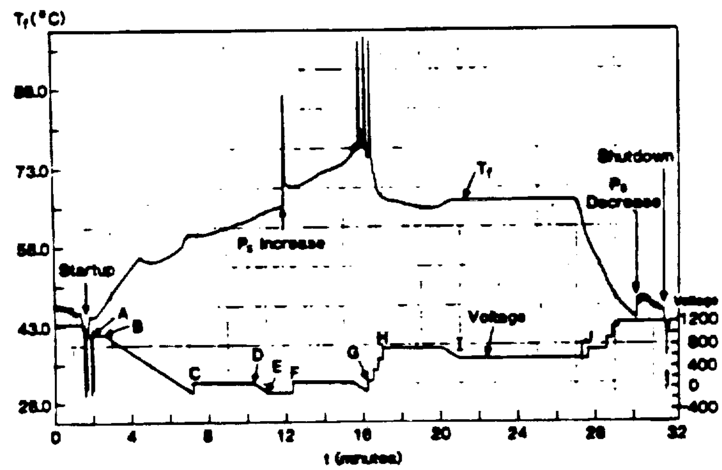

FIGURE 4. Transient performance, controllable industrial seal [1].

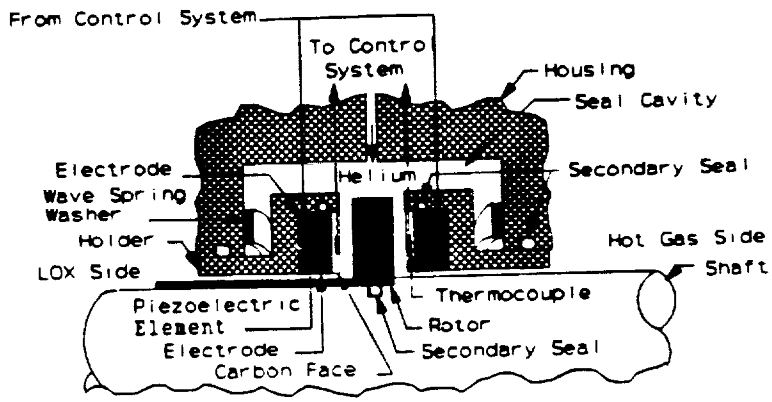

FIGURE 5. Controllable mechanical seal, aerospace application.

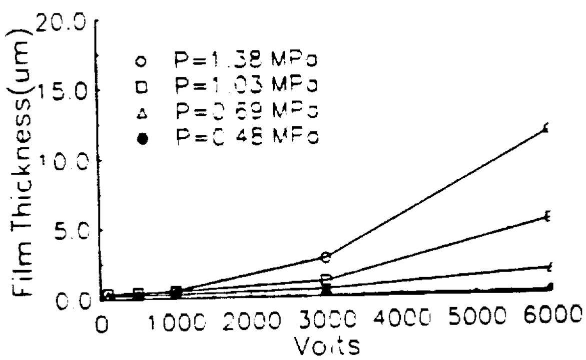

FIGURE 6. Predicted performance, controllable aerospace seal.